

Optimization of geometric features of circular saw blades and parameters of the manufacturing process aided by optiSLang

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Abstract

This article is dedicated to the numerical optimization of circular saw tools. Because of their geometrical shape, resembling a thin spinning disc, they are susceptible to vibrations and only possess a low static and dynamic stiffness. The main elements to achieve high-quality machining results with circular saw blades are their geometrical features and manufacturing parameters. The typical behaviour of these tools can be influenced by specific variation of the geometry of expansion slots and damping slots. On the part of the manufacturing process, a massive influence on the static and dynamic behaviour is exercised by the roll tensioning parameters. The target values of circular saw blades are high critical speeds and maximum static and dynamic stiffness. Previous attempts at optimizing and designing circular saw blades have been absolutely empiric and are not based on continuously parametrized models. This article shows in examples, that optimization using consequently parametrized models of circular saw blades is feasible. The calculated results are contrasted with and rated against conclusions of the latest state of technology. The results show, that the CAD-models of circular saw blades can be parametrized during the concept-phase using the presented approach and that building on these models, optimized products with high stiffness and optimum dynamic behaviour can be developed.

Keywords: Circular saw, optimization, roll tensioning, slots, natural frequencies, stiffness, critical speed

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1 Introduction

The initial stage of many industrial production processes is a sawing operation. Raw materials - such as a rod-shaped mass product for instance - will have to be cut to length before further machining in a modern production process of which many are highly or even fully automated. The economic efficiency of automated cutting processes directly correlates with their cutting performance. Thus, at high degrees of automation, circular saws offer a key advantage over other saw types such as band or hack saws. The importance of circular sawing is further exemplified by the ubiquity of small stationary circular saws or hand-held power tools used in workshops or in the craft trade sector for a multitude of materials ranging from wood or metal to plastics. Thus, circular sawing can be considered one of the most important preproduction processes. In a figurative sense, the circular saw tool establishes a link between machine and work piece. Since - just as with other production processes - the tool's mechanical properties ultimately define machining quality, the circular sawing process requires saw blades with optimized static and dynamic characteristics. This need is further emphasized by the blade's round, disc-like geometry and its centre mounting hole which makes it prone to oscillations and easily excitable by lateral force. The general tendency to smaller kerfs and therefore thinner tools places increased weight on optimization methods with which a blade's static and dynamic characteristics can be optimized. The spectrum of circular saw blades in the field of production technology is shown in Figure 1.



Figure 1: Spectrum of circular saw blades. Left: Machines for high output rates in mass-production (automation), Right: Battery driven handheld tool (Handcraft)

All these factors constitute a demand for research of optimization methods for circular saw blades. Both static and dynamic properties of a circular saw blade depend on several factors including the tool's rotation, its temperature profile, the slots and the stress condition of the blade due to roll tensioning. The influence of the thermal effects is not presented here.

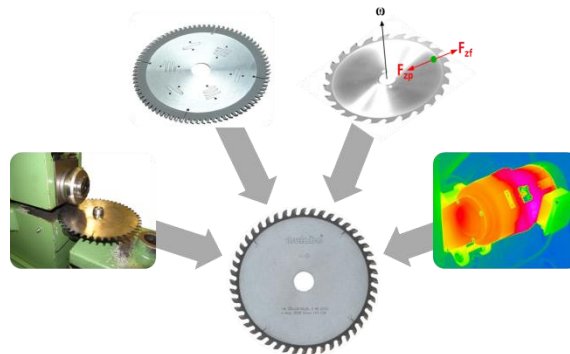


Figure 2: Main influences on saw blade dynamics and statics

A key part of this article focuses on the roll tensioning process which is one of the most crucial stages in the production of circular saw blades. Two rolls apply a contact pressing force and thereby plastically deform the blade (Figure 4). Internal residual stress, resulting from these deformations, improves the blades static and dynamic characteristics.

This method is well accepted throughout the scientific community, even though the underlying physics as well as its precise interrelation to the internal stress distribution are not yet fully disclosed. The influence of residual stress is successfully simulated using linear finite element models whereas conventional indirect models only describe the roll tensioning outcome, while lacking any information on precise stress distribution and actual roll tensioning process parameters. In [1] a model is presented which is able to calculate the real stress conditions within the tool using a nonlinear finite element model.

On another subject, this article covers expansion and damping slots which saw blades are usually equipped with in order to improve their static and dynamic behaviour. The number of slots necessary along with their shape and location is, in current industrial practice, commonly still being determined by empirical methods. In this work however, finite element calculations help identify significant slot parameters and allow quantifying their effects on blade performance. The presented algorithm can also be used to obtain an optimum slot configuration parameter set including slot length, orientation, location and quantity given certain base blade geometry.

Ultimately, this article provides simulation and optimization algorithms with which the relevant parameters of circular saw blades under conditions of operation can be predicted with a high level of accuracy. The results aim to give new impetus to the development, design and future advances of circular saw blades.

2 Theoretical background

2.1 Natural frequencies and critical speeds and stiffness of circular saw blades

2.1.1 Fundamentals

The global static and dynamic behaviour of circular saw blades can be approximately described by circular plates. The dynamic behaviour of a circular plate is characterized by nodal circles (n) and nodal diameters (k). Figure 3 gives some examples of characteristic mode shapes of a circular plate. As a general rule, mode shapes with few nodal diameters are most problematic for circular saws during operation. The statics and dynamics of a basic circular plate with no slots and not roll tensioning treatment can be solved using the Kirchhoff-Love theory [2,3], which can be considered a two dimensional bending line. Real circular saw blades with a complex stress state and slots cannot be calculated by such basic analytic approaches. Instead numerical solution approaches are necessary, like the finite-elements-method used in this article.

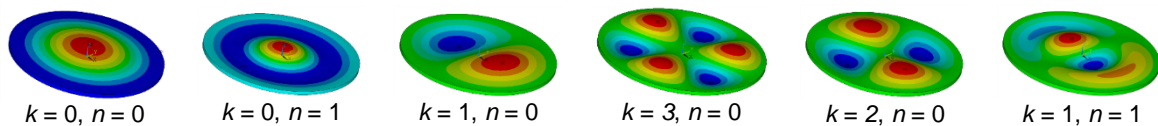


Figure 3: Characteristic mode shapes of a circular plate

2.1.2 Campbell-diagram and stability of circular saw blades

Long since the dynamic performance and the stability of spinning structures have been analysed. The authoritative works to this field of research were contributed by Southwell [4]. The fundamental result of his research is the fact, that natural frequencies of spinning plates can be deduced from natural frequencies of motionless blades. The centripetal and centrifugal forces which are affecting a spinning structure, cause stiffening of the structure and therewith an increase in eigenfrequencies in the stationary frame of reference. If speed dependant eigenfrequencies are plotted to a diagram, a frequency-speed diagram or Campbell diagram will be obtained. It describes the relation between eigenfrequencies of nonrotating saw blades in comparison to rotating ones. The so-called critical rotation speeds – critical speeds in short - can be extracted from the frequency-speed diagram. To illustrate the following circumstances a simple example of the frequency-speed diagram has been calculated and is shown in Figure 4.

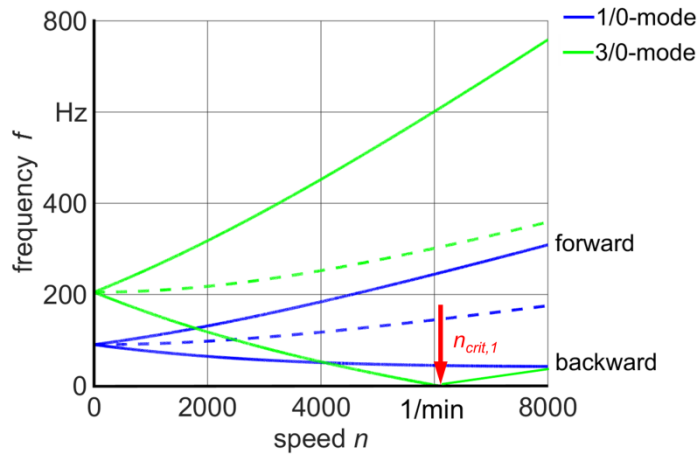


Figure 4: Schematic frequency-speed diagram (Campbell diagram)

In Figure 4 the dynamic circumstances for two mode shapes with one and three nodal diameters and no nodal circles are shown. To completely understand the frequency-speed diagram, the choice of reference systems is crucial. There are three lines plotted for each mode respectively. The so-called speed-dependant eigenfrequency as it would be recognized by a moving observer situated on the disc is represented by the dashed line in the middle. The two other lines for each mode, one of which is sloping up the other downwards, reveal themselves to an immobile observer, watching the blade.

Mathematically they can be deduced as the sum respectively the difference of the speed-dependant eigenfrequency and the product of the number of nodal diameters and the rotational speed according to Equation (3). According to [4] the speed-dependant eigenfrequencies can be approximated by the following analytical equation

$$f_{e\omega}^2 = f_e^2 + \lambda f_n^2 \quad (1)$$

In Equation (1) $f_{e\omega}$ denotes the rotation-speed-dependant eigenfrequency of a specific mode, f_e denotes the eigenfrequencies of the concerned mode of the immobile blade and f_n the rotational speed. The factor λ accounts for the mode shape and can be calculated using the number of nodal diameters k and the poisson ratio ν as followed:

$$\lambda = \frac{1-\nu}{4} \cdot k^2 + \frac{3+\nu}{4} \cdot k \quad (2)$$

The dimensionless poisson ratio ν is an elastic material property, which is the negative ratio of transverse to axial strain of a body. For metallic materials the value of ν is approximately 0.3, which is the value used for calculations in this paper.

It should be emphasized, that the approximation (2) is only valid for mode shapes with exclusive nodal diameters and no nodal circles. Equation **Fehler! Verweisquelle konnte nicht gefunden werden.** reproduces the true behaviour of unslotted circular saw blades satisfactory. If there are slots however, the factor λ has to be modified. In [1] this coefficient is quantified for annular plates and slotted plates.

Starting with Equation (1) the other two curves can be calculated as followed:

$$f_{1,2} = f_{e\omega} \pm k \cdot f_n \quad (3)$$

The curves can be imagined as a forward and a backward running wave on the blade. Both are moving waves of the same frequency and speed, though they are propagating in different directions. One wave moves in the direction of the rotation of the saw blade (f_1) the other

opposite to it (f_2). This results in the mode split shown in Figure 4. The positive term in (3) accounts for the forward whirl, the negative one for the backwards whirl. Using the Campbell diagram critical states can be detected.

The most important scenario resulting in instabilities of the saw blade can be read directly from Figure 4. The abscissa of the first intersection of a backwards whirl with the axis of the rotational speed is generally called the first critical speed $n_{crit,1}$ of the saw blade. At this rotational speed the circular saw blade loses its stability and an observer in the stationary frame of reference can register a standing wave with very large amplitudes. The blade seems to freeze in its shape. In this state very small lateral forces cause large deflections. This leads to a bad machining quality. To avoid this state of resonance in praxis, saw blade manufacturers usually specify the maximum permitted rotational speed with 0.7 to 0.85 times $n_{crit,1}$. This first critical speed is often caused by mode shapes without nodal circles and with only few nodal diameters. Increasing the first critical speed $n_{crit,1}$ is a fundamental aim of the optimization measures of circular saw blades next to the general elevation of their eigenfrequencies, maximizing their damping properties and static stiffness.

The critical speeds $n_{crit,k}$, measured in min^{-1} can be deduced from Equation (3) by postulating that the frequencies of backwards traveling waves have to be zero, i.e. $f_2 = 0$:

$$n_{crit,k} = \frac{60 \cdot f_{e,k}}{\sqrt{k^2 - \lambda}} \quad (4)$$

Here the speed-dependant eigenfrequency of a mode with k nodal diameters is represented by $f_{e,k}$.

2.1.3 Plate buckling and stability

Influenced by internal stress thin plates can show buckling effects similar to buckling columns [5]. If the internal stress exceeds a critical value this will lead to so-called plate buckling. In this state the circular saw blade loses its stability and it will be deflected perpendicularly to the plane of the plate. For circular saw blades buckling can be caused by unfavourable thermal loads or high roll tensioning forces. Caused by thermal influences saw blades tend to buckle in the 2/0 or the 3/0 mode shape. Excessive roll tensioning forces leads to buckling in the 0/0 or 1/0 mode. In this paper only buckling due to roll tensioning effects is taken into consideration. When the critical load case is reached the corresponding eigenfrequencies take on the value of zero. Assuming a linear-elastic behaviour it is possible to solve the eigenvalue problem, which allows to determine the critical loads and buckling mode shapes.

$$(\mathbf{K} + \xi \mathbf{K}_{SS})\mathbf{u} = \mathbf{0} \quad \text{Det}(\mathbf{K} + \xi \mathbf{K}_{SS}) = 0 \quad (5)$$

In these equations \mathbf{K}_{SS} represents the stress stiffness matrix or the geometrical stiffness matrix, which is described in detail in [6]. The matrix \mathbf{K} is the regular stiffness matrix and the vector \mathbf{u} stands for the displacements. The eigenvalue ξ is called load factor. The effective load has to be multiplied by this load factor to calculate the critical loads $F_{w,cr}$ and $T_{max,cr}$.

It has to be noted, that the calculated values are purely theoretical. Effects like structural imperfections, plasticities, contacts and geometric non-linearities are disregarded in this calculation. To take these into consideration a non-linear calculation has to be conducted. The critical tensioning forces $F_{w,cr}$ effective in this case can be determined by successively increasing the exerted loads using FE-analyses including the foresaid influences until the

structure reaches an unstable state. Because of imperfections, which in practice always exist, the critical loads calculated using the linear buckling analysis are only non-conservative values. Failure will occur sooner than calculated. This paper will only regard the linear buckling load analysis.

2.2 Roll tensioning

The natural resonance of the tool can be de-tuned by the internal stresses induced during roll tensioning, raising specific eigenmodes or lowering others. The critical speeds for modes with none or one nodal diameter are reduced; those with more than one nodal diameter are raised. The increase of the natural frequency is accompanied by an improvement of the static behaviour of the tool. Tensioning loads which are too high lead to buckling of the structure, similar to buckling columns. The fundamental process parameters of the roll tensioning procedure are tensioning force, the tensioning radius and the roll crown. Within this article only the influence of the tensioning radius is analysed. By application of the presented procedure optimal parameter values can be determined, based on given boundary conditions.

Based on the finite element method, the optiSLang optimization corroborates the findings of earlier works, which had proven a roll radius to blade diameter ratio of two thirds to be most appropriate. The optimization's objective function chosen in this work is the critical rotation speed. The optimization tool can be used to determine optimum values for the other parameters too.

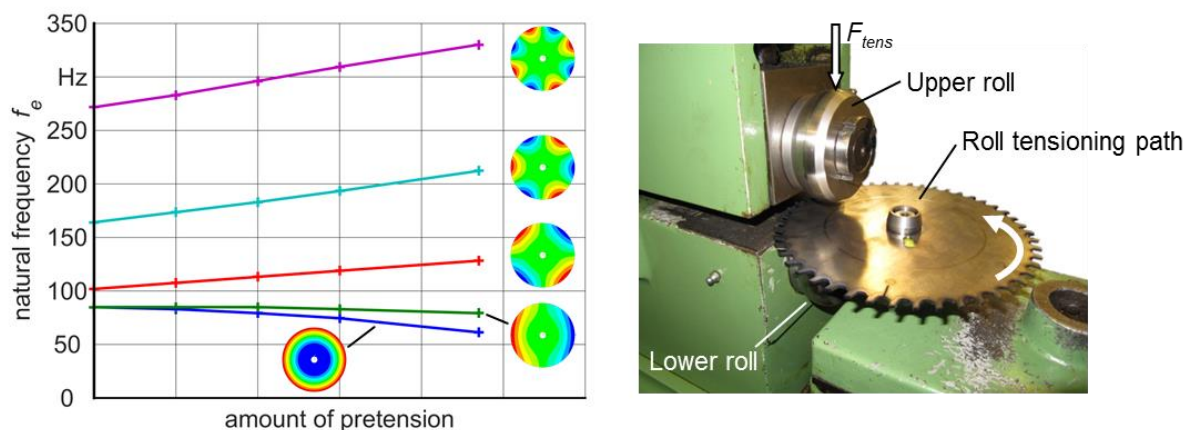


Figure 5: Natural frequencies dependent on the amount of pretension for different vibration modes (left), kinematics of the roll tensioning process (right)

The schematic progression of the eigenfrequencies, dependant on the tensioning force and the kinematics of roll tensioning process are presented in Figure 5. Countless papers exist, which describe the positive effects of roll tensioning on static and dynamic characteristics of circular saw blades. Among others the results are documented in [7-18].

2.3 Slots and circular saw blades

Experimental and analytic examinations of circular saw blades were often part of studies made in science and industry in further times. They often were accompanied by examinations of the effect of slots on static, dynamic and acoustical behaviour of circular saw blades [19-28]. Basically there are two types of slots: slots on the outer edge of the saw blade and slots in the inner area, the integration of slots at the edge counteracts the effect of thermal expansion, as the segments gain the possibility of tangential expansion. Two examples of the basic configuration of damping and expansion slots are shown in Figure 6. Slots can basically be oriented radially or tangentially on the saw blade or they can be designed in complex shapes and arbitrary directions as shown in Figure 6. Often viscoelastic materials or solid dampers, for example copper rivets, are integrated into the interior slots or the slot ends to increase the systemic damping.

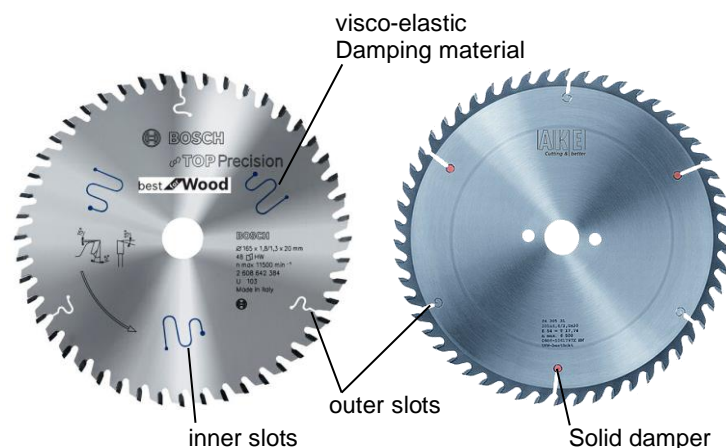


Figure 6: Slotted wood-cutting circular saw blades and attenuation medium (sources: Robert Bosch GmbH, AKE Knebel GmbH & Co. KG)

Even after many years of research, the modes of action aren't accurately recorded yet. The positive effect of slots is attributed to the interference to respectively the displacement of the natural frequencies of the saw blade. It was speculated, that structural weakening of the saw blade due to the slots allows for relative movement between its segments, which facilitates energy dissipation. Other interpretations assume that the reduction of internal compression stress at the outline of tool is the source of the positive effect of slots. Another positive impact is that standing waves are prevented at critical speed due to asymmetric conditions on the blade, which is established by integration of slots. Slots can reduce the oscillation amplitude because the vibrational energy is spread more widely in the spectrum. Many literary references point out, that because of the slot geometry eigenmodes may split into two distinct eigenfrequencies if the slot configuration fits the mode shape.

However literature also mentions that slots have a negative effect on the static stiffness, because they always weaken the structure. In context of optimization-problems, an optimum always represents a compromise between stiffness and dynamic behaviour. Finally, the analysed studies have shown that only a proper orientation of the slots leads to a raise of the critical rotation speeds, or a small minimization of the static stiffness respectively. One example for such a slot configuration is presented and analysed in chapter 3.1.3. Alongside

the many scientific studies in the area of slot geometries of circular saw blades, practical examinations were conducted by the industry, which point out the advantage of slotted over unslotted saw blades, for example [29] and [30]. There are also many independent developments and patent applications which exist on the part of the industry. An impression of the variations of slot geometries and positioning is given in Figure 7.

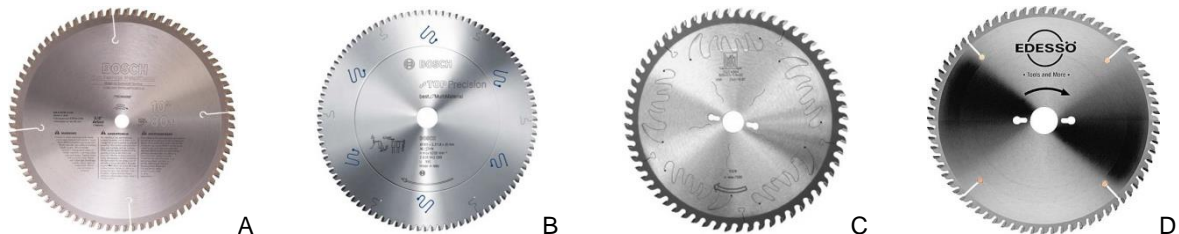


Figure 7: Examples of slotted circular saw blades (sources: Robert Bosch GmbH (A,B), Leitz GmbH & Co. KG (C), EDESSÖ Tools and More GmbH & Co. KG (D))

3 Optimization of circular saw blades: State of the art

3.1.1 Fundamentals

As was presented in the previous chapter, an extensive knowledge about roll tensioning and about slots is available. The significant modes of action aren't completely known yet, despite all the scientific work which was done by now. For this reason roll tensioning seems to be more like craftsmanship than a modern automated industrial production process. At almost all saw blade manufacturers roll tensioning and the handling of corresponding machines, is the task of long-time employees, which choose the process parameters under empirical aspects, building on their professional experience. For an outside audience roll tensioning as well as slot configuration seem like "alchemy". In most cases slot configurations are tailored to clients' requests based on experience values. For new applications and changed base conditions slot configuration are identified in empirical test series. The introduced results show, that modern calculation methods, accompanying the design process are able to generate optimized products. Using the suggested procedure, the engineer designing circular saw blades has a tool at his disposal, to automatically identify optimized slot configurations and process parameters for roll tensioning procedure.

3.1.2 Targets for optimization

Target values for optimized circular saw blades are manifold. In general a high static stiffness for circular saw blades is to be pursued. Especially in metal working, tools are exposed to great loads because of high machining forces. Therefore an increased stiffness at the tools outer rim is needed. Founded in the much lower rotation speeds in comparison to wood-working, a high critical rotation speed is not mandatory for metal working tools. To be able to tolerate the prevailing high temperature gradients the endurable buckling loads have to be as high as possible. This is true for both metal machining as well as wood-working, as high temperatures can occur at the teeth. Usually circular saw blades are designed to be fatigue-endurable. Especially in the slotted areas the tension has to be minimized, by choosing

appropriate slot geometries. Ultimately it is obvious, that circular saw blades are exposed to complex stresses and strains which necessitate a multi-objective optimization.

In the following part exemplary calculations are made for the optimization of circular saw blades. A sensitivity analysis and basic problems relating to the roll tensioning process are defined and analysed. The chosen procedure is presented and the results are compared with actual conclusions and measurements.

3.1.3 Example 1: Sensitivity analysis of a slot configuration

The first example will show how a sensitivity analysis can be applied to identify the relevant geometric parameters of an exemplary slot configuration on the most important technical properties of the saw blade base body. The slot configuration used for these calculations is based on the works of [31]. The geometry of the blade and its slots is depicted in Figure 1.

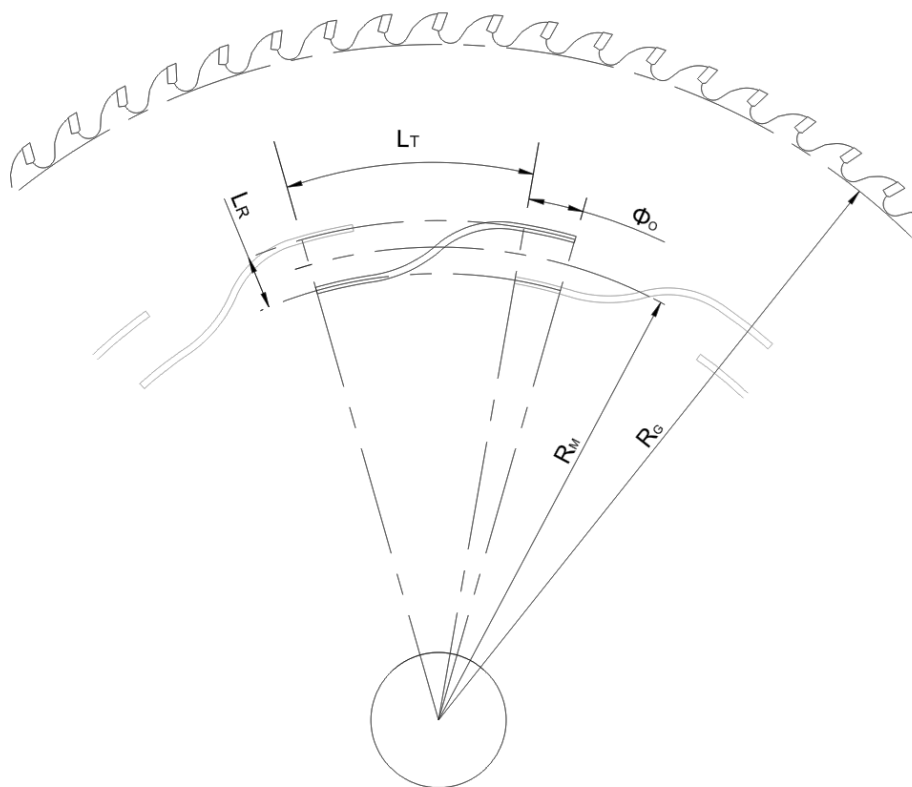


Figure 8: Slot configuration in compliance with [31]

The blade base and the slot configuration are defined by the tooth base radius R_G , the mean slot diameter R_M , the length of the circular arc L_T of a single slot without the overlap ϕ_o , the radial slot length L_R as well as the slot width s and the blades thickness t . In compliance with the cited patent the lengths respectively radii are combined into the dimensionless aspect ratio Q_L (11) and the radius or diameter ratio Q_D (12).

$$Q_L = \frac{L_T}{L_R} \quad (6)$$

$$Q_D = \frac{R_M}{R_G} = \frac{D_M}{D_G} \quad (7)$$

For the sensitivity analysis the aspect ratio Q_L and diameter ratio Q_D as well as the slot width s and the overlapping angle ϕ_o are defined as model parameters. The ranges of the parameter values are set to allow only feasible configurations, to prevent merging of slots for example. The blade diameter is constant at 300 mm as are the blade thickness (2.2 mm) and the number of slots (14).

As explained earlier there are several objectives that are relevant to optimizing saw blades:

- The static stiffness c_{st} . The ratio of a lateral force F , applied to the outer edge of the blade, to the corresponding displacement w . It represents the blade's resistance to transverse disturbances (8).
- The maximum equivalent stress σ_v at the ends of the slots.
- The margin between the rotational frequency of the blade and its closest eigenfrequency Δf_{min} . This parameter substitutes for the critical speed, as the saw blade in this scenario is intended for a specific rotational speed ($n_0 = 3000 \text{min}^{-1}$) (9).
- And the sum of the first 10 eigenfrequencies at operational speed Δf_{sum} , as a characteristic for the blades overall dynamic behaviour (10).

$$c_{st} = \frac{F}{w} \quad (8)$$

$$\Delta f_{min} = \min \{f_0 - f_{n,i}(n_0)\} \quad (9)$$

$$\Delta f_{sum} = \min \sum_{i=1}^{10} f_{n,i}(n_0) \quad (10)$$

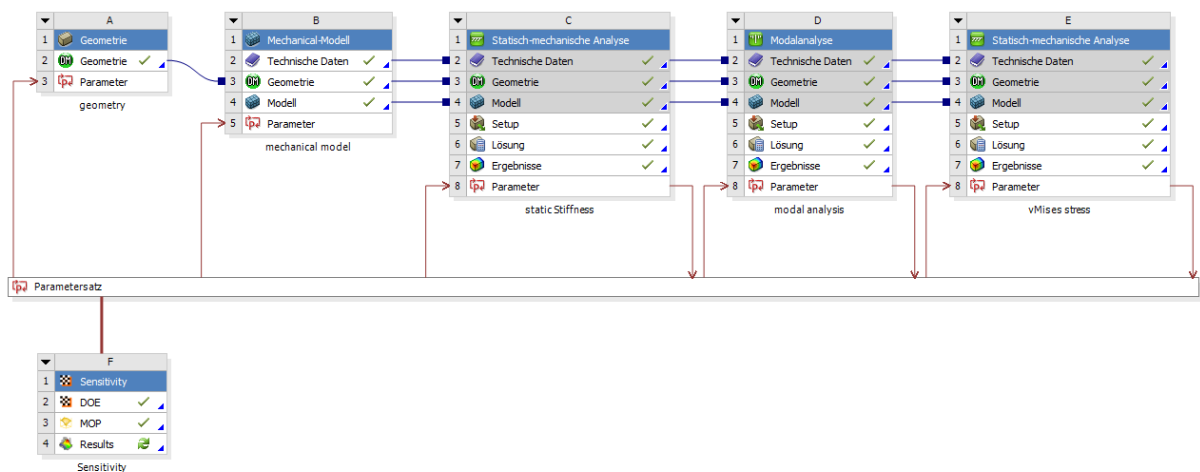


Figure 9: Workflow in ANSYS Workbench

The FE-simulation used to calculate the objective function follows the workflow shown in Figure 9. APDL scripts in each block distill the objective functions out of the separate analyses.

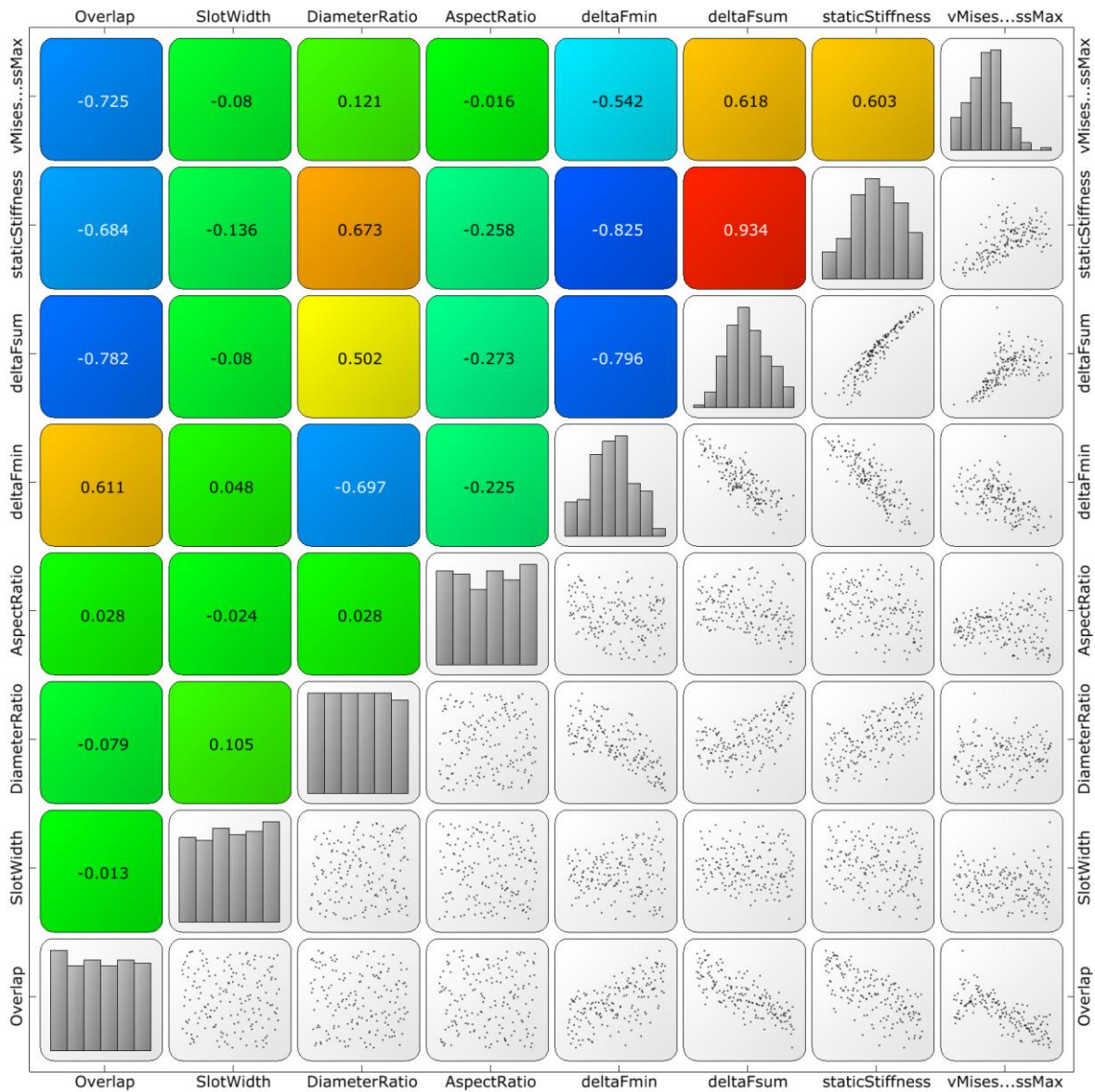


Figure 10: Extended correlation matrix

In Figure 10 row-wise opposite algebraic signs of the correlation coefficients show that the operation specific objective Δf_{\min} and the more general objectives c_{st} and Δf_{sum} are conflicting with each other. The contrary trends of static stiffness and frequency margin of the blade won't allow finding an optimum slot configuration for every application. Depending on the application of the saw blade (wood-working or metal machining) the objectives have to be weighted differently in future optimizations.

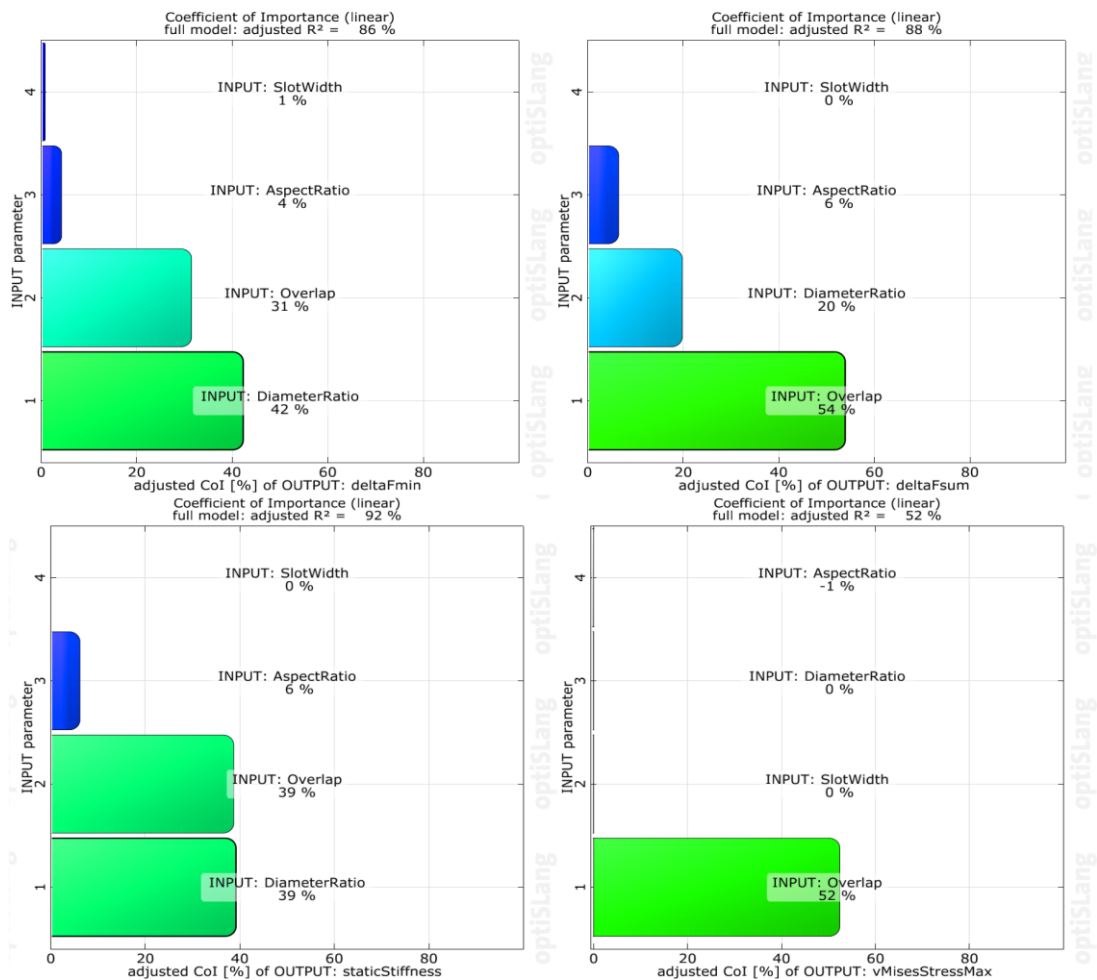


Figure 11: Coefficients of importance

Representations of the coefficients of importance for each objective reveal at first glance that the slot width and the aspect ratio seem to have little impact on either of the objectives (Figure 11). This will allow future optimizations to omit these parameters in order to save computing time.

3.1.4 Example 2: Optimization of roll tensioning radius

The first example didn't consider the effects based on roll tensioning. This example will show that it is possible to identify ideal tensioning parameters by consequently parameterizing the model. The roll tensioning process is incorporated in ANSYS using a linear prestress analysis. Within the Workbench environment this is realised by an APDL-Script. The residual stress effects are modelled via different thermal expansion coefficients in the area of the roll tensioning path and the rest of the blade. For these procedures the optimization parameter is the roll tensioning temperature T . This is a non-mechanical substitute for the physical tensioning force F . The tensioning temperature can be converted to the physical domain "force" by aid of corresponding characteristic curves. The chosen approach delivers qualitative information concerning the magnitude of the pretension.

This example will determine the radius r_t which offers a maximum the first critical speed. This optimization problem is typical for wood-working applications. When calculating optimal tensioning radii the effect described in chapter 3.1.3 has to be considered. Tensioning forces or temperatures $T_{t,crit}$ which are too high lead to buckling of the saw blade, so tensioning temperatures higher than $T_{t,crit}$ should be eliminated in advance. For this examination a procedure was chosen, which identifies the maximum permissible tensioning temperature corresponding to a particular radius in an initial calculation. These critical temperatures $T_{t,crit}$ are transmitted to the optimizer for every design m via the parameter manager of ANSYS. The typical approach is illustrated in Figure 12. This procedure rules out, that invalid, physically unfeasible values are given to the optimizer. In this scenario the parameter is no independent parameter, but rather a range for variations between its lower bound and the maximum temperature for each respective design m . For the implementation of this approach, an auxiliary variable WB_{help} has been established, for which a range of values between 0 and 1 was defined. This value is committed to Excel, where the temperature value $T_{max,crit,m}$ for the current radius is interpolated.

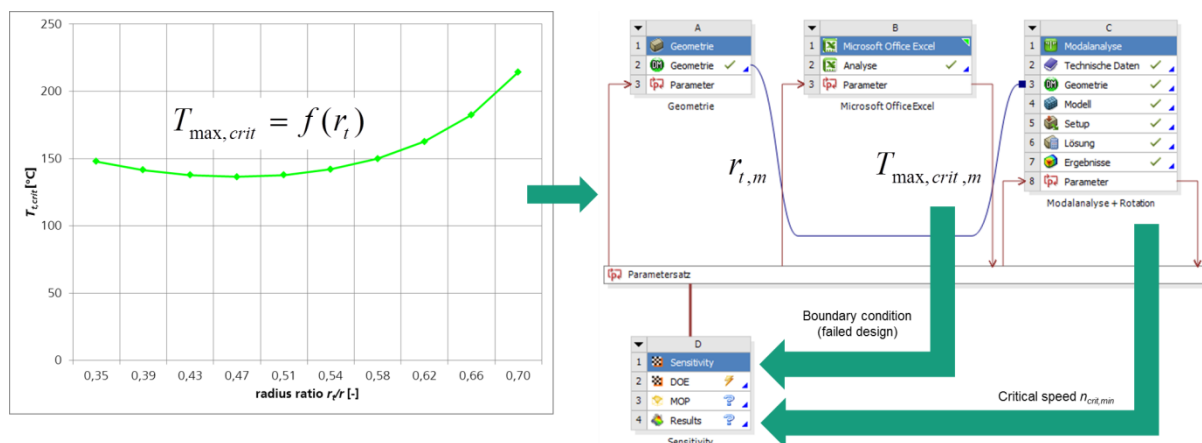


Figure 12: Basic procedure of the initialization step for the roll tensioning optimization

This yields the following procedure:

1. Variation of the parameters WB_{help} und $r_{t,m}$ for the current design m by optiSLang within the global range and transfer to Excel
2. Computation of the valid maximum value $T_{max,crit,m}$ on basis of $r_{t,m}$.
3. Calculation of the temperature for the design m and transfer of the value to the parameter manager for further use in ANSYS:

$$T_t/T_{min} \rightarrow WB_{help} \leftarrow T_{max,crit,m} \cdot T_{min} \quad (11)$$

The minimal temperature was set to a constant value. Roll tensioning can lead to transposition of modes. For this reason the eigenmodes have to be tracked using an automated process. As shown in Equation (11) the knowledge of nodal diameters is necessary for the identification of mode shapes and the establishment of the Campbell diagram. Mode tracking and the extraction of nodal diameters and circles is performed by an APDL-script. The basic procedure of building a Campbell diagram is shown in Figure 13:

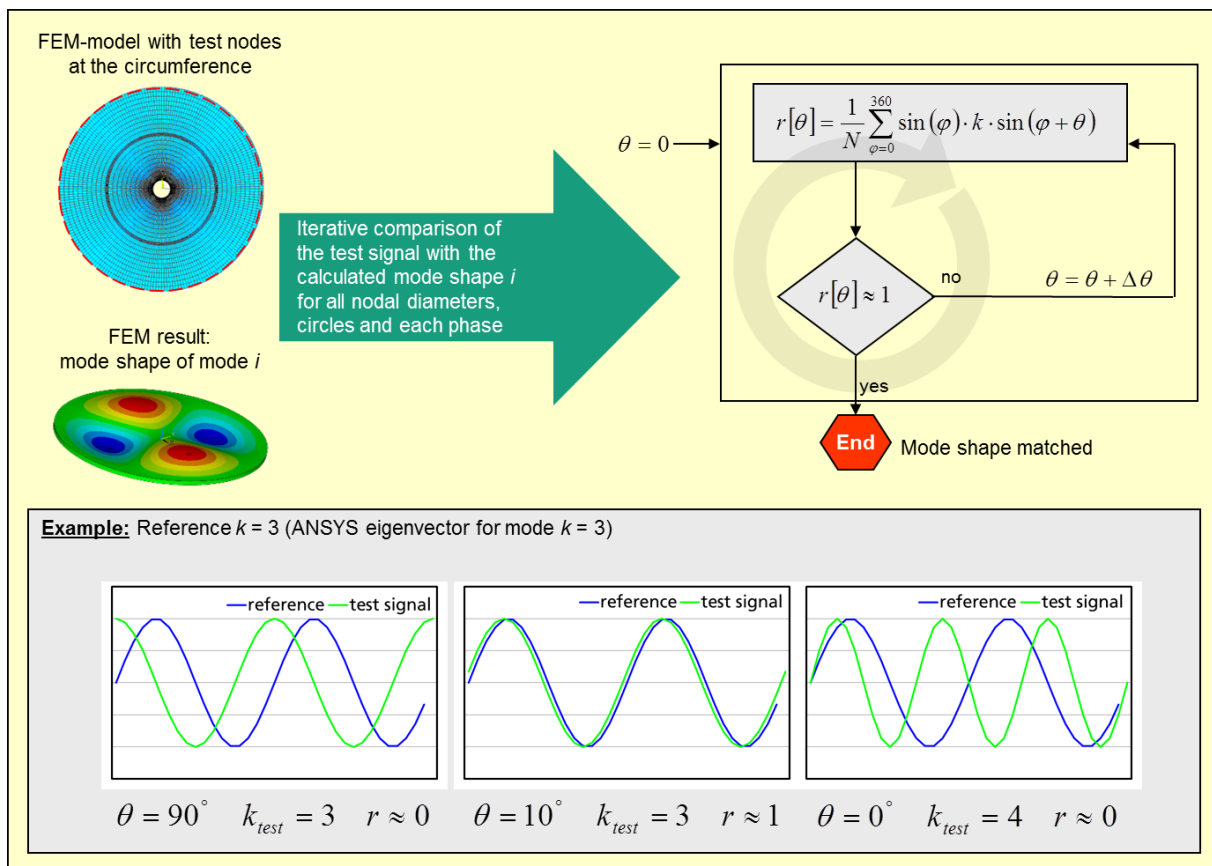


Figure 13: Basic procedure of the implemented mode tracking algorithm and example based on the 3/0-modeshape

The knowledge, that mode shapes of circular saw blades are made up of nodal circles and nodal diameters, which can be represented as a combination of analytic sine waves, is used for mode tracking. An analytic test model is generated, within which nodal diameters and circles are known. This model is compared to the FE-model of the circular saw blade. At the outer edge of the circular saw blade the nodal solutions of the FE-model are selected for a specific eigenvector i . Then they are compared to the analytical signal.

$$r[\theta] = \frac{1}{N} \sum_{\varphi=0}^{360} \underbrace{\sin(\varphi) \cdot k}_{\text{reference (FEM) modeshape } i} \cdot \underbrace{\sin(\varphi + \theta) \cdot k}_{\text{analytical test signal}} \quad (12)$$

By shifting the test signal over the phase angle for all possible combinations of nodal diameters and circles correlation values for the current configuration are generated, which measure the conformity of the modes. By iteratively comparing the analytical test signal with each calculated mode shape of the FE-model for every nodal diameter and circle the identity of the eigenmode and its shape can be determined. Correlation values near to 1 indicate a match, values of 0 a mismatch, as shown in Figure 13.

The results of the optimization run are contrasted to experimental data and experience values. In practice circular saw blades are roll tensioned at a radius r_t at 2/3 of the outer diameter. This ratio is independent of the tool geometry, but varies depending on the flange, which

clamps the blade in its center. To identify the eigenmodes an experimental modal analysis was conducted. The tools were analysed under free-free boundary conditions. Annular plates were used, which were roll tensioned at varying diameters. The experimental setup and the results are depicted in the following figure. The outer diameter of the plates is 175 mm, the central bore 30 mm and the plate's thickness 2.4 mm

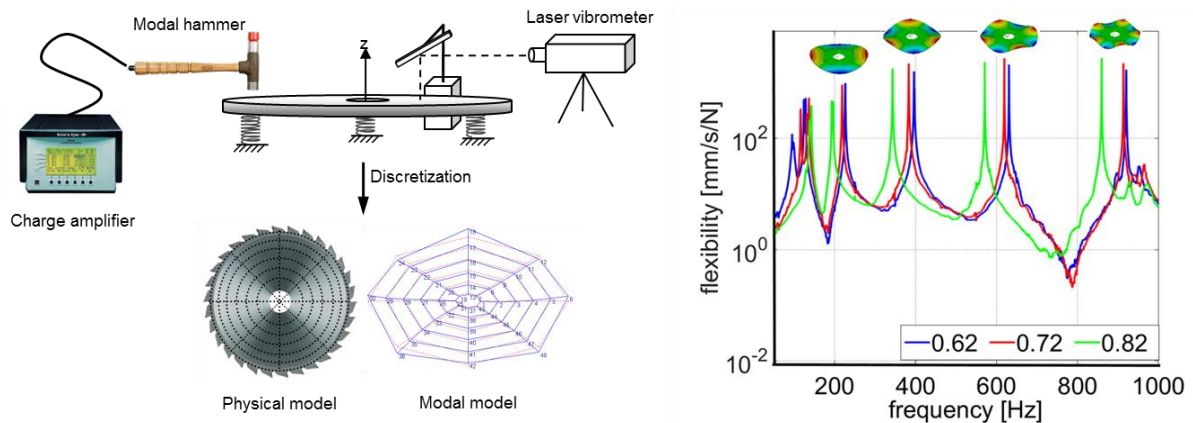


Figure 14: Measurement setup and frequency response functions of circular plates with different roll tensioning radii

Figure 14 shows clearly, that the highest eigenfrequencies of the relevant eigenmodes are attained for a ratio of tensioning radius to outer radius of 0.62. Extrapolating the results by use of Equation (3) to the domain of the rotational speed, the highest critical speeds are attained at this ratio as well. As a substitute for a fully detailed circular saw blade the optimum tensioning radius of an annular plate is calculated with optiSLang using the previously described calculation procedure. The results are depicted in Figure 15. The geometry of the plate is in accordance with the experimental setup. Up to a diameter of 110 mm the inner area of the plate is clamped, to account for the flange. The degrees of freedom of the corresponding nodes are locked.

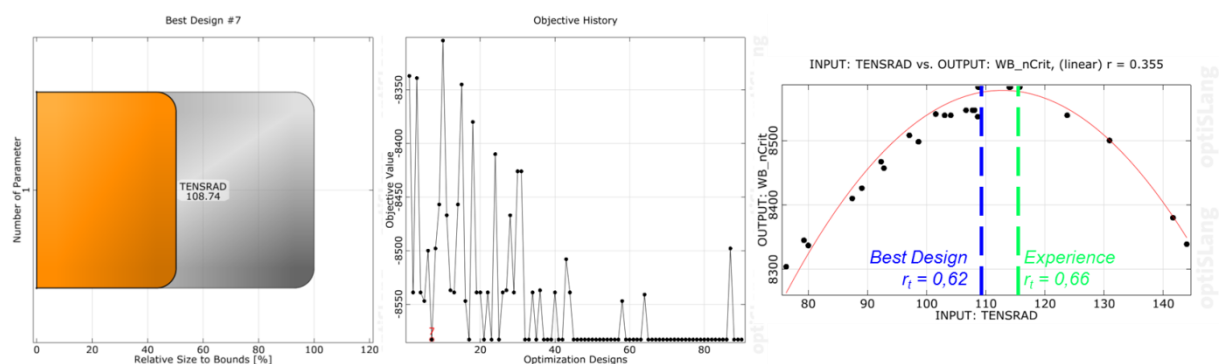


Figure 15: Calculation results (Optimization of the roll tensioning radius)

The left-hand representation in Figure 15 shows the parameter and its bounds. An optimum tensioning radius of 109 mm was calculated. Divided by the outer diameter of 175 mm this results in a ratio of 0.62, which conforms to the experimental data shown in Figure 15 and the educated guess use in practice. The middle representation depicts the objective function

which is converging quickly and clearly after only a few designs. The characteristic quadratic curve of the right-hand graph in Figure 15, which shows the critical speed as a function of the tensioning radius, compares well to literary sources, i.e. [9,10].

4 Conclusions

By means of concrete practical application it was proven that circular saw blades can be improved using parametric optimization techniques. The results concerning tensioning parameters and slot configurations show a good match with conducted experiments, empirical values and the state of technology. In future the analyses will be expanded to further applications. Especially the examination of rotating tools during a machining process. To add this feature and more to the procedural method will be supplemented by the involved research institutions. With the sketched method the design engineer eventually will have a simple tool to analyse and refine circular saw blades during the design process.

5 References

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